Design, Optimization and ANSYS analysis of component of ATV wheel assembly.

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ABSTRACT: In automobile industry it is essential to produce the light weight assembly in order to increase the vehicle performance. Also, lots of forces during braking, acceleration and bump conditions are also applied directly during dynamic condition. So, the project is concerned with design, analysis and manufacturing of knuckle and hub. The objective of this paper is to reduce weight and increase performance of vehicle. This paper deals with calculation of various loads and their simulation. We also have used the lotus to find the hard point of the knuckle to improve the steering geometry as well as the suspension geometry of vehicle. The FEA result indicates that the upright assembly is able to perform safely in real track condition as per performance requirement.

It must also be noted that the components must be designed in such a way that they have a minimum weight at the same time care must be taken that they do not cross a certain limit of stress value. In this Project, the Complete Design Procedure of the Wheel Assembly for AT-111 Rims with Tires (23*7*10) has been presented along with the optimization of the same components. The weight of the vehicle is considered to be 300 kg along with the driver. The project illustrates the forces acting on the components, the failure criteria, and the optimization of the components. The project deals with finding out the dimensions of the individual components and also detecting the probable regions of stress concentration. The design procedure follows all the rules laid down by FMAE Rule Book for Quad bike design challenge

Keywords: Dynamic terms, design, load transfer, wheel assembly, ANSYS analysis, knuckle, hub.

INTRODUCTION

There are always two types of masses in an automobile – sprung and unsprung mass. All the mass of the vehicle that is damped by the spring is called the sprung mass. As the Wheel Assembly mass is not damped by the spring, it comes into the unsprung mass category. We know that the unsprung mass must be lower than the sprung mass and also should be as least as possible to provide proper drive stability and load balancing of the vehicle. Thus, it becomes important to reduce the mass of the wheel assembly and the rims and tires. But while doing this care must be taken that the mass of the wheels, tires, and the wheel assembly must be enough to prevent the lateral toppling of the vehicle at the time of cornering or impact.

There are a lot of forces acting on the wheels in the static and especially in the dynamic condition. As the Wheel assembly is directly connected to the wheels, all these forces also have an impact on the designing of the Wheel Assembly. A lot of forces act on the wheel assembly during accelerating, braking, cornering, and tilting.

A good Wheel Assembly is one that can sustain such forces over a longer period. Thus, it is required to design the wheel assembly considering all these factors. A failure of any component of the Wheel Assembly means a breakdown of the automobile and in some cases might also be hazardous for the driver. Thus, utmost care must be taken while designing the Wheel Assembly. The objective of Optimization is always to find the best possible and suitable dimension. This is because optimization does not always mean reducing dimensions it also means finding out the dimensions which will be just enough to sustain the force.

1. Design methodology

1.1 Steps involved in methodology

Step 1: Modelling of steering knuckle using 3D modelling software.
Step 2: Finite element modelling of the steering knuckle.
Step 3: Analysis of steering knuckle using ANSYS software.
Step 4: Finite element stress analysis.
Step 5: Modal analysis

1.2 Problem statement

Automobile vehicle required having higher speed and power, its steering knuckle has higher strength and stiffness but must be lighter in weight and size.

In developing power output vehicles, importance is placed on the weight of the linear and angular parts steering arm, tie rod, etc.

The overall performance of the steering system is affected by higher inertia forces, generated by the moving parts of the vehicle. Therefore, it should always be investigated to avoid any failure of the vehicle in the long run.

1.3 Why it is important

Due to uneven stress distribution over steering knuckles, its life reduces. So, it is dangerous for the rider of the vehicle while off roading and also results in additional maintenance cost.
This affects the overall performance of an automobile vehicle. Due to the lack of knowledge of stress distribution the material wastage has occurred. It also increases overall weight of the vehicle and is larger in size than required size.

1.4Aims and objectives

The aim of this project is to optimize the performance of automobiles by reducing the weight and removing stress acting on it by designing and analysis. According to the calculation using the right material to reduce size and weight.

2. Design of Spindle

Firstly, the spindle is designed on which other components such as knuckle, bearings and hub will be fitted. At this stage we cannot decide the actual length of the spindle, so we just consider the tentative length of the spindle.

Material: - The material for manufacturing the spindle is taken to be EN24. There will be parts which will be press fitted on the spindle. So, heat treatment will be necessary to increase surface hardness. Besides the yield strength in tension of EN24 is also high.
Syt = 654 N/mm2
Endurance Limit = 412 N/mm2

2.1 Determining the forces acting on the spindle:

The forces acting on the spindle are as follows

a. Weight of the vehicle

During static and dynamic conditions, a constant force of the self-weight is acting on the spindle at the part inside the knuckle. Even if it is considered as the more than half the weight of the car is acting at the front portion of the car during braking, the weight on the one wheel is

Weight in the front portion = 104kg

Weight on one tire= $\frac{104}{2} = 56\ kg$

Force = 56× 9.81 = 550 N

Let us consider this weight to be **700N.**

b. Bump force on the tire

At the time of a bump in the surface a force will act on the portion of the spindle which is inside the spindle. This is because the hub is bolted directly to the wheel. This force is obtained from the wheel rate. For design purpose the wheel rate is kept as 45N/mm2. Also, it is considered that there will be no bump more than 30mm as the track is extremely flat.

Bump Force $= 3G = (3 \times 9.81 \times 56) = 1650\ N$

Consider, Bump Force = **1700N**

c. Torque on the spindle.

Torque = mass on the spindle × g × radius of the wheel × coefficient of friction

= 60× 9.81 × 0.292× 0.7 = 120 N

Torque on spindle = 130 N

2.2 Designing and determining the dimensions.

Shear Failure of the Spindle

The spindle is likely to fail in shear because of the bump force. The spindle is a critical part and it is not at all desirable to fail any condition, hence the factor of safety is taken to be 3.

The allowable shear stress$= \tau = \frac{S_{yt}\times 0.5}{F.O.S}$

$$= \frac{654 \times 0.5}{3} = 109\ \ N/mm^2$$

Now,

$$\tau = \frac{Force}{Area}$$

$$109 = \frac{1700}{\frac{\pi}{4} \times d^2}$$

Therefore, d=5mm

Thus, from the shear failure the diameter of spindle 'd' comes out to be 5 mm

Bending and Torsional Failure of the Spindle

To find out the Maximum Bending Moment, the SFD and BMD are to be drawn. Referring to the force diagram of the spindle,

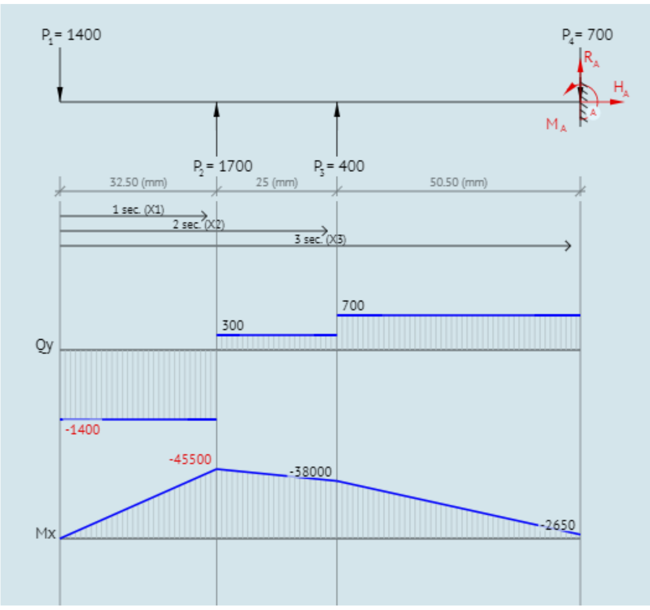

Fig Shear force and bending moment diagram

Equating forces along Y-axis to 0

Ra + Rb =700-1700

Ra + Rb = - 1000 N

Equating the moments about point

A =0

- (1700 × 32.5) – (Rb × 57.5) + (700 × 50.5) = 0

Rb=346N

Rb = 400N

Ra + 400N = - 1000 N

Ra = -1400N

Ra = -1400N

After reconsidering the force diagram comes out be as shown in figure Now,

Moment about P1= 0

Moment about P4= (700 × 50.5) = 35350 N-mm

Moment about P3= (700 × 75.5) – (400× 25) = 42850 N-mm

Moment about P2= (700 ×108) – (400 ×57.5) – (1700 × 32.5) = -2650 N-mm

From above it is clearly seen that the maximum bending moment is at point P3,

Thus Mc= 42850 N-mm

The Torque acting on spindle can be directly taken from result number **Mt= 130 N-m = 130000N-mm**

As EN24 is a ductile material, using maximum shear stress theory to find out the diameter of the spindle.

$$d^3 = \frac{16}{\pi \times \tau} \times \sqrt{(M_b^2 + M_t^2)}$$

$$d^3 = \frac{16}{\pi \times 109} \times \sqrt{(42850^2 + 130000^2)}$$

Therefore, d=18.56mm=20mm

Thus, from the bending and torsional failure the diameter of spindle 'd' is 20 mm

3. Design of Knuckle

Knuckle is that part of the wheel assembly which is press fitted on the spindle and the A-arms are also mounted on the Knuckle. Besides the knuckle also serves the function of providing mounting to the Brake Caliper. The Steering Arm which is used to connect the wheel assembly and the tie rod is also mounted on the knuckle. Thus, due to all these mountings, there are a lot of forces acting on the knuckle. The Knuckle as such is subjected to completely reversed types of stress while turning from one turn to the other and also during braking and accelerating. Thus, taking a material called EM24. The material properties are as follows: -

Syt = 654 N/mm2

Endurance Limit = 412 N/mm2

3.1 Determining the forces acting on the Knuckle:

The forces acting on the spindle are as follows

a. Longitudinal Forces during Braking: -

While Braking, the weight of the rear side tends to come in the front side of the vehicle so there is a load transfer that is taking place form rear to front. It interns affects the knuckle as these forces act on the A-arm mounting points through the A-arms.

Considering Maximum acceleration of 1g = 9.81 m/s2

Force at the front side = mass at the rear side of the vehicle × acceleration

Let the mass at the rear side of the vehicle be 0.6 times the total weight

Therefore,

Mass at the rear side of the vehicle=0.6 × 300 =180 kg

Force = 180 × 9.81 Force = 1765.8 N

Now force on 1 wheel =1765.8/2 =882.9 N

Thus, Longitudinal Force =882.9 N

b. Lateral Forces: -

Let the vehicle take a turn of 6m turning radius and at a speed of 30kmph

r = turning radius =6m

v= 30 kmph=8.333m/s

Centrifugal Force = m*v^2/r

= 0.4*300*8.333*8.333/6

=1388.77 N

Now consider if all the weight at the front side comes on the wheel assembly the force will be Force due to lateral load transfer =0.4 × 300 × 9.81 =**1175.5 N**

Force acting up on each brake mount = (2759.06/2) = **1379.5N.**

c. Force on the Steering Arm: -

According to the steering effort, the force on the steering arm was found out to be 1165.52 at an angle of 8.5560. After resolving these forces.

Force 1 = 1165.82 × cos (8.556) = 1152.84 N

Force 2 = 1165.82 × sin (8.556) = 173.44 N

But the force on steering arm =1165.52 N

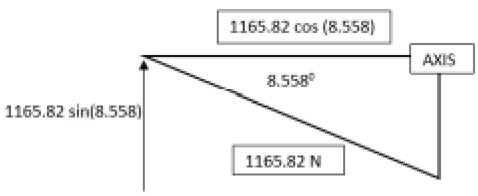

Fig. Steering Arm Forces

d. Forces on the caliper mounting points due to torque

The radius for the upper and lower caliper mount points are 89.13 mm and 60.233 mm. The maximum force will be at minimum radius. Hence consider the force on the lower arm.

Force (lower) = torque / radius = 130000/60.23 = 2158.28 N

$$shear\ stress\ on\ bolts(\tau) = \frac{S_{yt} \times 0.5}{factor\ of\ safety}$$

$$\tau = \frac{580 \times 0.5}{2}$$

$$\tau = 145\ N/mm^2$$

The force acting on the caliper bolt is given by the result (Vi)

Now,

$$\tau = \frac{F}{A}$$

$$\therefore \tau = \frac{2158.23}{(\frac{\pi}{4} \times d_c^2)}$$

$$\therefore d_c = 4.3533\ mm$$

$$d = \frac{d_c}{0.8}$$

$$d = \frac{4.3533}{0.8}$$

$$d = 5.44\ mm$$

Thus, selecting the caliper bolt size as M8.

3.2 Selection of Knuckle Upper and Lower Bracket Bolt

The bolts are standard parts and have a defined value of yield strength. All bolts used in the Wheel Assembly are made up of a minimum of 8.8 Grade.

Syt = 580 N/mm2

Factor of Safety = 2

Shear Force acting on these bolts = $\sqrt{(442 * 442) + (695 * 695)}$= 823.64N

Shear Stress on the bolts= $\tau = \frac{S_{yt} \times 0.5}{F.O.S}$

$$\tau = \frac{580 \times 0.5}{2}$$

$$\tau = 145\ N$$

now,

$$\tau = \frac{F}{A}$$

$$145 = \frac{884}{2 \times (\pi r^2)}$$

dc = 1.90mm

d = dc/0.8 = 2.37mm

This value is too small. Thus, for practical reasons selecting the bracket bolt size as **M8.**

3.3 Longitudinal Shear Failure of the Knuckle Bracket.

$$Allowable\ stress\ in\ the\ knuckle\ in\ shear = \tau = \frac{S_{yt} \times 0.5}{F.O.S}$$

$$\tau = \frac{654 \times 0.5}{2}$$

$$\tau = 164\ N$$

Longitudinal force acting on bracket = 884 N

now,

τ =force/area

$$\tau = \frac{F}{A}$$

$$164 = \frac{884}{t \times b}$$

$$t \times b = 5.4$$

Where, t= thickness of bracket

b= distance between the hole and the end of bracket

If t = 2.9mm b= 1.9 mm

Thus, for practical reasons the width of bracket in longitudinal direction is taken to be 18mm

The thickness of the bracket is taken as 12 mm

3.4 Lateral Shear Failure of the Knuckle Bracket

Allowable stress in the knuckle in shear

= τ = Syt*0.5 / fos =654*0.5/2 = 164N

$$Allowable\ stress\ in\ the\ knuckle\ in\ shear = \tau = \frac{S_{yt}}{F.O.S}$$

$$\tau = \frac{654 \times 0.5}{2}$$

$$\tau = 164\ N$$

Lateral force acting on bracket = 1390 N

now,

$$\tau = \frac{F}{A}$$

$$164 = \frac{1390}{t \times b}$$

$$t \times b = 8.5$$

Where,

t= thickness of bracket

b= distance between the hole and the end of bracket

If t = 3.6mm b= 2.4 mm

Thus, for practical reasons the width of bracket in lateral direction is taken to be 15 mm

The thickness of the bracket is taken as 10mm

3.5 Bending Failure of Knuckle

Bending due to Longitudinal Force

This bending is due to the force of 883 N. The longest part of knuckle is 86 mm away from the center and the knuckle is almost symmetric. Thus bending moment in such cases is taken to be

Mb = 883 × 0.5 × 86

Mb = 37969 N-mm

Now,

By Flexural Equation,

$$\frac{M_b}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{S_{yt}}{F.O.S}$$

$$\sigma_b = \frac{654}{2}$$

$$\sigma_b = 327\ \ N/mm^2$$

$$y = \frac{b}{2}$$

$$I = \frac{1}{12} \times t \times b^3$$

Where,

t= thickness of knuckle, b= width of knuckle

$$\frac{37969}{\frac{1}{12} \times t \times b^3} = \frac{327}{\frac{b}{2}}$$

$$\frac{37969 \times 12}{t \times b^3} = \frac{327 \times 2}{b}$$

$$t \times b^2 = 696.67$$

If t=14mm, b = 7mm

But here it is also important to understand that the spindle will be fitting in the knuckle thus for this reason the width of the knuckle is taken as 50mm at the center and would then decrease to 40 mm till the end.

Thus, the width of the knuckle is 50mm.

The thickness of knuckle is 16 mm

3.6 Bending due to Centrifugal Force

This bending is due to the force of 1388.77 N. The longest part of the knuckle is 86 mm away from the center and the knuckle is almost symmetric.

Thus, bending moment in such cases is taken to be

Mb = 1388.77 × 0.5 × 86

Mb = 59717.11 N-mm

Now, By Flexural Equation,

$$\frac{M_b}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{S_{yt}}{F.O.S}$$

$$\sigma_b = \frac{654}{2}$$

$$\sigma_b = 327 \ N/mm^2$$

$$y = \frac{b}{2}$$

$$I = \frac{1}{12} \times t \times b^3$$

Where,

t= thickness of knuckle, b= width of knuckle

$$\frac{59717.11}{\frac{1}{12} \times t \times b^3} = \frac{327}{\frac{b}{2}}$$

$$\frac{59717.11 \times 12}{t \times b^3} = \frac{327 \times 2}{b}$$

$$t \times b^2 = 1091$$

If, t=17 mm, b = 8 mm

But here it is also important to understand that the spindle will be fitting in the knuckle thus for this reason the width of the knuckle is taken as 50mm at the center and would then decrease to 40 mm till the end.

Thus,

The width of the knuckle is 50mm.

The thickness of the knuckle is 16 mm.

3.7 Shear Failure of the Steering Arm

$$Allowable\ stress\ in\ the\ knuckle\ in\ shear = \tau = \frac{S_{yt} \times 0.5}{F.O.S}$$

$$\tau = \frac{654 \times 0.5}{2}$$

$$\tau = 164\, N/mm^2$$

Force acting on Steering Arm as obtained from result (v) is 1165 N

now,

$$164 = \frac{1165}{2 \times (t \times b)}$$

t × b = 4.6322

Where,

t= thickness of steering arm

b= distance between the hole and the end of arm

If t = 6mm b= 0.77 mm

Thus, for practical reasons the width of steering arm is taken to be 6 mm

The thickness of the bracket is taken as 10 mm

3.8 Bending Failure of Steering Arm

This bending is due to the force of 1165 N. The longest part of steering arm is 50 mm away from the center and the knuckle

Thus, bending moment in such cases is taken to be

Mb = 1165 × 50

Mb = 58250 N-mm

Now, By Flexural Equation,

$$\frac{M_b}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{S_{yt}}{F.O.S}$$

$$\sigma_b = \frac{654}{2}$$

$$\sigma_b = 327 \ N/mm^2$$

$$y = \frac{b}{2}$$

$$I = \frac{1}{12} \times t \times b^3$$

Where,

t= thickness of steering arm in contact with knuckle

b= width of steering arm in contact with knuckle

$$\frac{58250}{\frac{1}{12} \times b \times t^3} = \frac{327}{\frac{b}{2}}$$

$$\frac{58250 \times 12}{b \times t^3} = \frac{327 \times 2}{t}$$

$$b \times t^2 = 1091$$

If t=16mm b = 8mm

Thus,

The width of the steering arm is 8mm.

The thickness of the steering arm is 16 mm.

3.9 Shear Failure of the Caliper Mounting

$$Allowable\ stress\ in\ the\ knuckle\ in\ shear =$$

$$\tau = \frac{S_{yt} \times 0.5}{F.O.S}$$

$$\tau = \frac{654 \times 0.5}{2}$$

$$\tau = 164\, N/mm^2$$

Force acting on Steering Arm as obtained from result is 2158.28 N

Now,

$$164 = \frac{2158.28}{(t \times b)}$$

$$t \times b = 13.16$$

Where,

t= thickness of caliper mount

b= distance between the hole and the end of mount

If t = 6mm b= 2.86 mm

Thus, the width of the caliper mount is taken to be 8 mm

The thickness of the bracket is taken as 6 mm.

4.0 Design of Hub

Hub is the part of wheel assembly on which the wheel and disk are mounted. Both the Wheel as well as the disk are mounted on the hub with the help of bolts. As discussed earlier the outer race of the bearing is press fitted in the hub and hence provision is made in the hub to enclose the bearing. The Hub itself is made of 2 Petal parts. One of the wheels and the other of the brake disk

4.1 Determining the forces acting on the Hub: -

The following Forces are acting on the Hub.

a. Torque on the Brake Disk Petal: -

A torque of 130 Nm is acting on the Brake Disk Petal.

The force acting on each hole =

$$= \frac{\left(\frac{moment}{radius}\right)}{no\ of\ holes} = 880\ N$$

b. Torque on the Wheel Petal: -

In order to sustain this braking effect, the wheel must also provide and equal and opposite torque. Thus, the magnitude of torque is same but the direction is opposite

The force acting on each hole =

$$= \frac{\left(\frac{moment}{radius}\right)}{no\ of\ holes} = 879.86\ N$$

c. Force due to Side Impact: -

If the vehicle is banged by other vehicle from side or if the vehicle has a collision with the fencing from side, there are chances that the petals might bend. Hence this side impact force must also be considered.

Here the side Impact force is taken to be 2G = 2 × g × vehicle mass

Therefore, Impact force = 2 × 9.81 × 300 = 5886 N

$$Impact\ force\ on\ 1\ petal = \frac{8829}{3} = 1962\ N$$

d. Loads on Bearing: -

The load on 1 bearing is 700 N

The load on 2nd bearing is 634 N

The axial load on the bearing is 1388 N

4.2 Selection of Wheel Bolt

$$Shear\ stress\ on\ the\ bolts = \tau = \frac{S_{yt} \times 0.5}{F.O.S}$$

$$\tau = \frac{580 \times 0.5}{2}$$

$$\tau = 145\ N/mm^2$$

The force acting on the Wheel Bolt is given by the result

Now,

$$\tau = \frac{F}{A}$$

$$\therefore 145 = \frac{880}{\frac{\pi}{4} \times d_c^2}$$

$$d_c = 3\ mm$$

$$d = \frac{d_c}{0.8}$$

$$d = \frac{3}{0.8} = 3.75\ mm$$

Also, in the rim there is a provision of M12 Bolt. Hence selecting bolt of M12.

4.3 Selection of Brake Disk Bolt

$$Shear\ stress\ on\ the\ bolts = \tau = \frac{S_{yt} \times 0.5}{F.O.S}$$

$$\tau = \frac{275 \times 0.5}{2}$$

$$\tau = 68.75\ N/mm^2$$

The force acting on the Wheel Bolt is given by the result

Now,

$$\tau = \frac{F}{A}$$

$$\therefore 68.75 = \frac{879.89}{\frac{\pi}{4} \times d_c^2}$$

$$d_c = 4.03\ mm$$

$$d = \frac{d_c}{0.8}$$

$$d = \frac{4.03}{0.8} = 5.0375\ mm$$

Thus, selecting the bolt size of **M8**

4.4 Design of Wheel Petal

Shear Failure of Petal

$$Allowable\ shear\ stress\ in\ the\ hub = \tau = \frac{S_{yt} \times 0.5}{F.O.S}$$

$$\tau = \frac{503 \times 0.5}{2}$$

$$\tau = 125.75\ N/mm^2$$

Force acting on Petal as obtained from result is 880 N

Now,

$$\tau = \frac{F}{A}$$

$$\therefore 125.75 = \frac{880}{(2 \times t \times b)}$$

$$t \times b = 3.5$$

Where,

t= thickness of Wheel Petal

b= distance between the hole and the end of petal

If t = 8mm, b= 0.475 mm

Thus the width of the petal is taken to be 7mm

The thickness of the petal is taken as 8 mm.

4.5 Bending of Wheel Petal

This bending is due to the force of 880 N. The radius of effective bending is 49.25 mm

Mb = 880 × 49.25

Mb = 43340 N-mm

Now,

Now, By Flexural Equation,

$$\frac{M_b}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{S_{yt}}{F.O.S}$$

$$\sigma_b = \frac{503}{2}$$

$$\sigma_b = 251.5\ N/mm^2$$

$$y = \frac{2b}{2}$$

$$2b = d$$

$$I = \frac{1}{12} \times t \times d^3$$

Where,

t= thickness of knuckle

b= width of knuckle

$$\frac{43340}{\frac{t\times d^3}{12}} = \frac{251.5}{\frac{d}{2}}$$

$t \times d^3 = 1034$

If t= 8mm d = 10.1 mm

Thus, the width of knuckle is 12mm.

The thickness of knuckle is 8 mm.

Total thickness is = width + diameter of hole =12+14 = 26 mm

4.6 Design of Brake Disk Petal

Shear Failure of Petal

$$Allowable\ shear\ stress\ in\ the\ hub = \tau = \frac{S_{yt} \times 0.5}{F.O.S}$$

$$\tau = \frac{503 \times 0.5}{2}$$

$\tau = 125.75\, N/mm^2$

Now, By Flexural Equation,

$$\frac{M_b}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{S_{yt}}{F.O.S}$$

$$\sigma_b = \frac{503}{2}$$

$\sigma_b = 251.5\ N/mm^2$

$$y = \frac{t}{2}$$

$$I = \frac{1}{12} \times b \times t^3$$

t= thickness of petal

b= width of petal =27 mm as found from above calculation

Force acting on Petal as obtained from result (vii) is 880 N

Now,

$$\tau = \frac{F}{A}$$

$$\frac{49540.5}{\frac{27\times t^3}{12}} = \frac{251.5}{\frac{t}{2}}$$

t = 6.61mm

The thickness of petal is taken as 8 mm.

5. Design of component

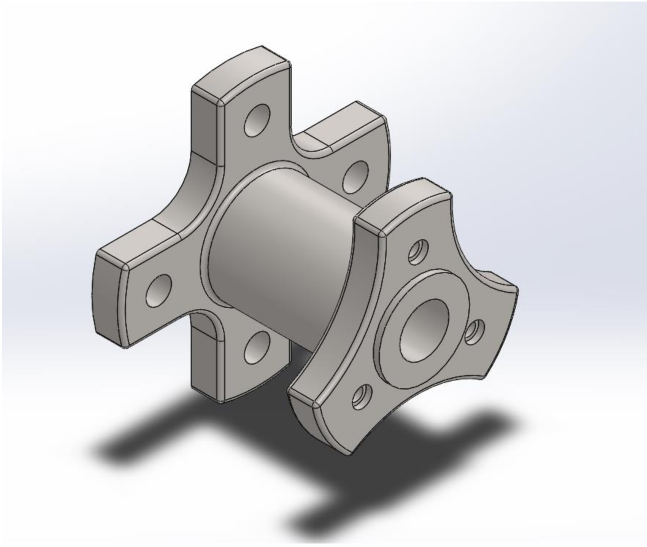

Fig. 5.1 Hub CAD Model

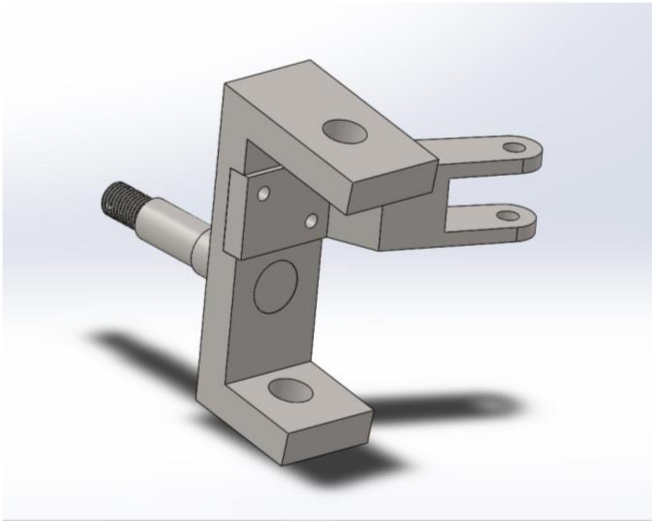

Fig 5.2 Knuckle CAD Model

Fig 5.3 Wheel Stud

Fig 5.4 Allen Bolt

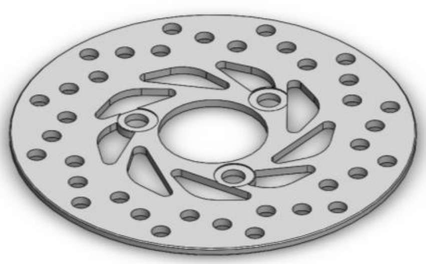

Fig 5.5 Disc

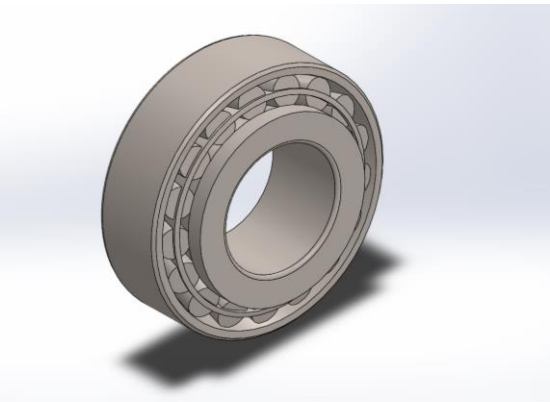

Fig 5.6 Taper Roller Bearing

6. Analysis and Optimization

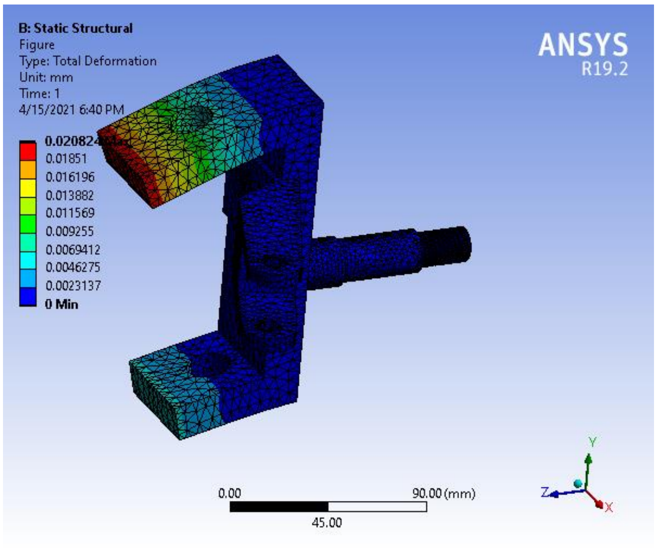

FIGURE 6.1
Model (B4) > Static Structural (B5) > Solution (B6) > Equivalent Elastic Strain

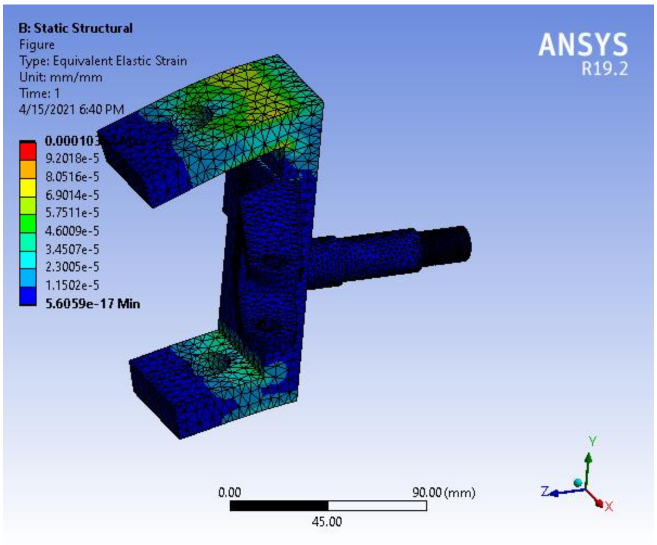

FIGURE 6.2
Model (B4) > Static Structural (B5) > Solution (B6) > Equivalent Stress

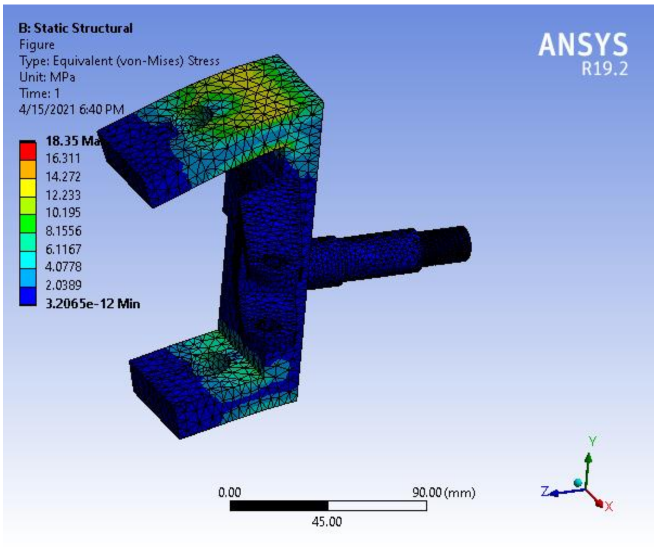

FIGURE 6.3
Model (B4) > Static Structural (B5) > Solution (B6) > Strain Energy

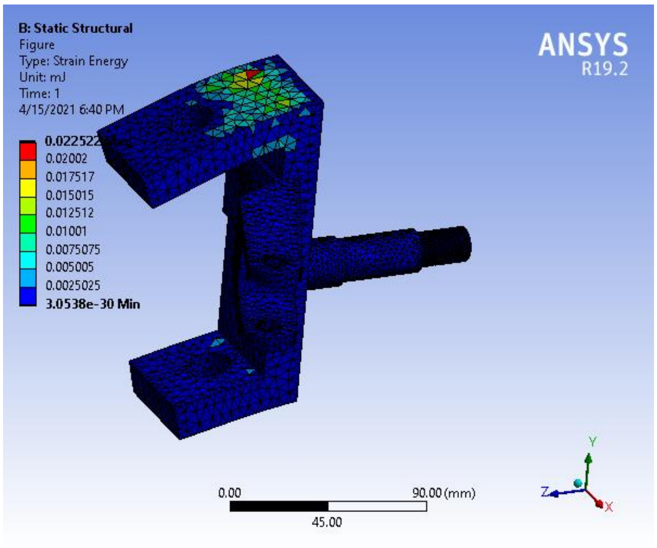

FIGURE 6.4
Model (B4) > Static Structural (B5) > Solution (B6) > Strain Energy > Figure

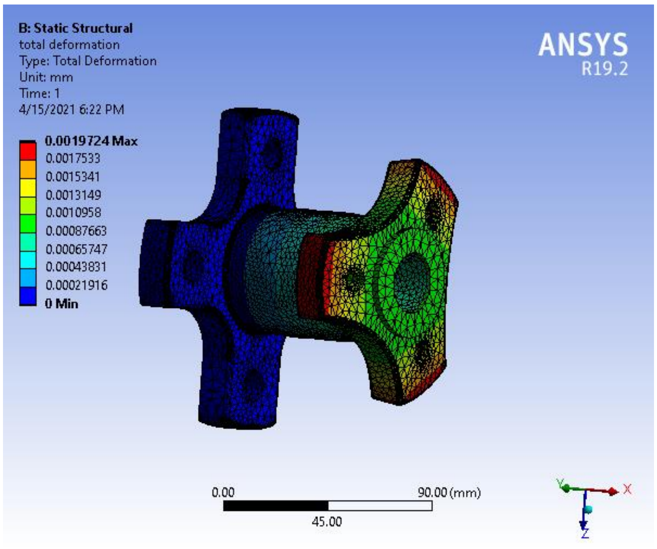

FIGURE 6.5
Model (B4) > Static Structural (B5) > Solution (B6) > Total Deformation > total deformation

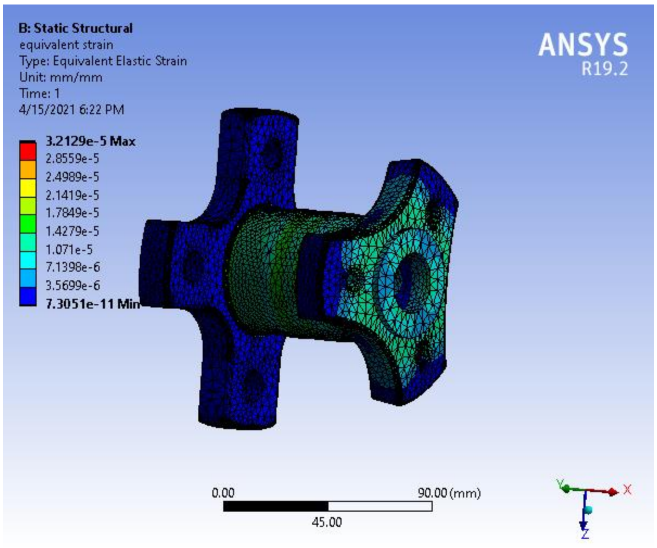

FIGURE 6.6
Model (B4) > Static Structural (B5) > Solution (B6) > Equivalent Elastic Strain > equivalent strain

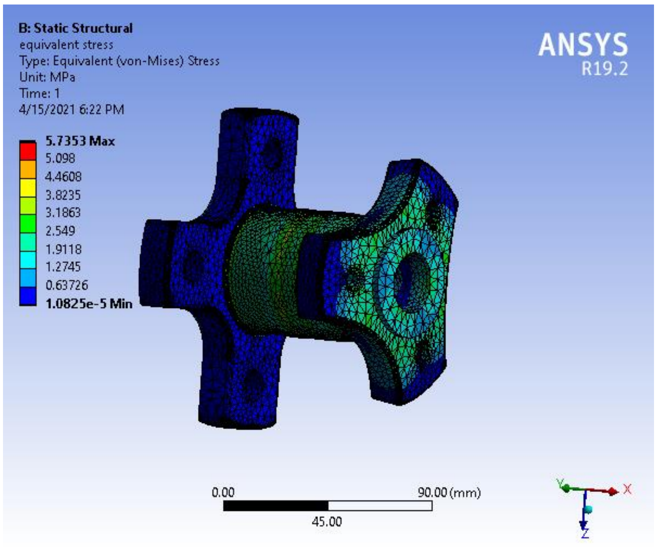

FIGURE 6.7
Model (B4) > Static Structural (B5) > Solution (B6) > Equivalent Stress > equivalent stress

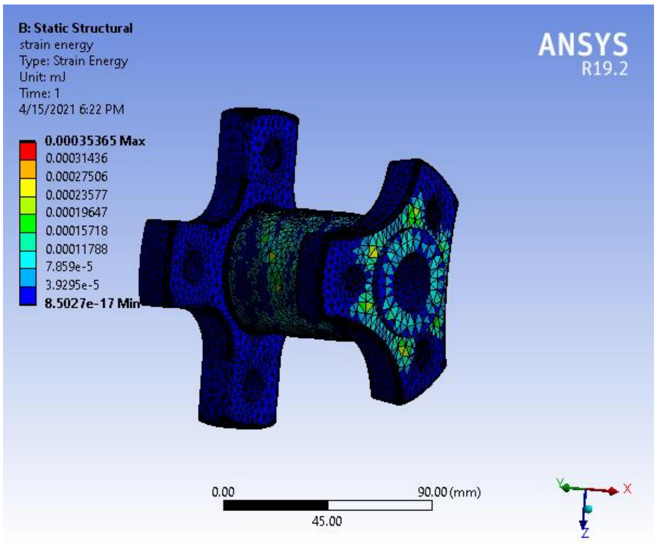

FIGURE 6.8
Model (B4) > Static Structural (B5) > Solution (B6) > Strain Energy > strain energy

7. Manufacturing

By studying various material properties, for front hub, for knuckle and steering arm we selected EN24.

No .	Part	Material	Process	Time
1.	Hub	EN24	CNC turning, vertical surface milling.	4
2.	Knuckle	EN24	Vertical machining and surface milling	6
3.	Steering arm	EN24	Milling	2

8. Result

The Knuckle and Hub has been modelled and analysed using Ansys. The various parameters such as displacements, Stress distribution are completely analysed and studied. This study shows that the areas where the stress concentration is maximum due to the applied load and the portions that has to considered in the design of steering knuckle in order to avoid frequent failures to improve its reliability. Therefore, design is safe

9. Conclusions

The purpose of this project is not only to design and manufacture the upright assemblies for the car, but also to provide an in-depth study in the process taken to arrive at the final design. With the overall design being carefully considered beforehand, the manufacturing process being controlled closely, and that many design features have been proven effective within the performance requirement of the vehicle. The FEA result indicates that the upright assembly is able to perform safely in real track condition as per performance requirement.

10. References

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